PATENT SPECIFICATION

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PROVISIONAL SPECIFICATION

Improvements in or relating to Hydraulic Reciprocating Pumps and Motors

I, FREDERICK HENRY CAREY, a British subject, of 5, Newbold Terrace, Leamington Spa, Warwickshire, do hereby declare the nature of this invention to be as follows:—

The present invention relates to small hydraulic pumps and motors intended primarily for use on aircraft for operating various devices and mechanisms thereon, although they may be used for any other purposes for which they may be found useful.

In the case of aircraft, for example, hydraulic pumps are used for creating hydraulic pressure for operating the rams of hydraulic jacks or hydraulic motors for actuating retractable under-carriages, flaps, gun turrets, and the like, whilst hydraulic motors are used for operating various mechanisms associated with retractable under-carriages, flaps, gun turrets, variable pitch air screws, and so on.

The object is to provide a highly efficient motor or pump which operates at very high pressure, and the invention is more particularly concerned with pumps and motors of the swash-plate type, in which the cylinders and pistons are arranged axially in a circle about the driving shaft, with the ends of the pistons bearing against or connected to the swash-plate.

The invention relates more particularly to the valve control for the inlet to and discharge from the cylinders. According to the invention the flow of liquid to and from the cylinders is controlled by a valve disc which rotates with the shaft and is provided with separate liquid passages and ports, whereby during rotation of the disc the cylinders are successively brought into communication as required with the liquid outlet and inlet connections of the machine.

Preferably the valve disc is disposed between the end wall of the cylinder block, containing ports communicating with the cylinders and the end cover of the machine carrying the liquid inlet and outlet connections, and the said valve disc is surrounded by an annular distance piece or ring enclosing and sealing the space between the cylinder end wall and the end cover. In said valve disc are separate arcuate annular grooves or slots in the surface adjacent to the cylinder end wall, on the same pitch circle or circles as the cylinder ports, and a hole is bored through the valve disc to provide communication between one of said slots and a continuous circular groove in the face of the end cover which is constantly in communication with the liquid inlet connection. The other annular slot or groove communicates by a radial bore with a continuous peripheral groove round the edge of the valve disc, and a passage in the distance piece or ring provides constant communication between this groove and the liquid outlet connection of the end cover.

The arcuate extent and relative positions of the two grooves or slots in the valve disc determine the valve timing, which is such that the inlet arcuate groove is over the cylinder ports of the cylinders whose pistons are performing a suction stroke, whilst the other arcuate groove is over the cylinder ports of the cylinders performing a power or delivery stroke, the connections of the various cylinders to inlet and outlet constantly progressing round the circle of cylinders as the swash-plate operating the pistons rotates. Each cylinder will be in communication with the inlet and outlet connections respectively during the whole of its piston travel, for both the suction and delivery strokes, thus contributing to high volumetric efficiency.

Various modifications of construction are desirable according as to whether the machine is to be used as a pump or as a motor, and these modifications will now be briefly indicated.

In the case of a pump the inlet and outlet grooves or slots in the valve disc will both have the same pitch circle corresponding to that of the ports of the cylinders which serve both for inlet and outlet, whilst to prevent overheating during long periods of blow-off pressure a liquid relief valve is housed in the end cover, to permit relief of the liquid to a connection on the end cover, from which it can be

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conducted to the supply tank or container.

Owing to the high delivery pressure which is desirable, and in order to obtain high volumetric efficiency, it is important that the working clearances between the valve disc, cylinder end wall, and end cover, be reduced to a minimum, but in order to minimise surface friction, parts of the contact surfaces which are not subject to delivery pressure, for instance the area on one face of the valve disc between the inlet groove and the shaft, and the whole centre area of the end cover adjacent to the inlet groove are preferably relieved.

This relieving renders it possible to provide in a simple manner for automatic lubrication of the complete working parts of the pump. The lubricant can flow from the relieved area of the end cover to a drilling in the shaft along which it flows to the centre of the pump. By providing a spiral groove in the surface of the driving shaft the lubricant can be returned towards the valve disc, to the relieved area thereof, thus completing the circuit.

According to a modification, instead of using a separate distance ring surrounding and enclosing the valve disc, the distance piece may be integral with the end cover of the pump. It will be understood, of course, that the joints between the distance piece, the cylinder end wall, and the end cover, must make a tight seal and for this purpose are preferably lapped. Also these parts are located in relation to one another by dowelling and secured as one piece to the cylinder block by bolts.

In the case of a motor the valve disc may be substantially the same as for a pump if rotation will only be required in one direction, but to permit rotation in either direction the grooving of the valve disc and the porting arrangements must be modified, as hereinafter explained, owing to the solid areas which must necessarily exist between the ends of the arcuate inlet and outlet slots on the cylinder side of the valve disc, and which serve to provide the necessary seals between the inlet and outlet grooves, between which a considerable pressure difference exists.

These sealing areas shut off each cylinder from both inlet and outlet ports at the two ends of the stroke, but owing to the non-compressibility of the liquid it is essential that each cylinder be in communication with the delivery connection at the commencement of the delivery stroke in the case of a pump or with the exhaust connection at the commencement of the exhaust stroke in the case of a motor.

In the case of a pump the piston load on the swash-plate is in one direction on the delivery stroke and in the reverse direction on the suction stroke, whilst in the case of a motor the piston load for both the power and exhaust strokes is in the same direction. Consequently the foregoing requirement may be met, in the case of a pump, by providing for a lost motion connection between the pistons and the swash-plate, which allows a sufficient angular movement of the valve disc to permit the sealing areas to pass the cylinder ports whilst the pistons are at their dead centre position.

An alternative method in the case of a pump would be to make the delivery groove in the valve disc a full 180° arc and to reduce the arc of the suction groove to provide sufficient intervening sealing areas, so that inlet or suction would not continue throughout the whole of the suction stroke. In the case of a non-reversible motor a similar arrangement may be used, the 180° groove being the exhaust groove. Such an arrangement would, however, not be suitable for a reversible motor in which the ports must act alternatively as both inlet and exhaust ports, according to the direction of rotation.

In the case of a reversible motor, therefore, in order to prevent attempted compression of the liquid, the grooves in the valve disc should preferably be arranged at different pitch circles, permitting each groove to be a full 180° arc, with radial sealing areas separating their ends. This involves providing each cylinder with two ports radially spaced correspondingly to the grooves in the valve disc.

Alternatively, instead of having grooves in the valve disc at different pitch circles, they may be of equal arcuate extent (somewhat less than 180°) and the cylinders may be provided with a pair of ports spaced circumferentially as far apart as possible, the solid areas between the ends of the grooves in the valve disc being substantially of the same width as the distance between the cylinder ports. This arrangement will permit momentary direct connection of the inlet with the exhaust via the two cylinder ports, but this will be for so short a period that the efficiency of the motor will not be affected to any serious extent.

In the case of a non-reversible motor the same automatic lubricating system can be employed as with a pump, but for a reversible motor such a system is not possible owing to the absence of the relieved surfaces on the valve disc and end cover. In this case arrangements may be made to maintain the

whole motor body full of lubricating liquid.

It is, of course, not necessary in the case of a motor to provide a relief valve in the end cover.

Dated this 8th day of June, 1938.
CHARLES S. PARSONS,
Chartered Patent Agent,
Thanet Houes,
231, Strand, London, W.C.2.

COMPLETE SPECIFICATION

Improvements in or relating to Hydraulic Reciprocating Pumps and Motors

I, FREDERICK HENRY CAREY, a British subject, of 5, Newbold Terrace, Leamington Spa, Warwickshire, England, do hereby declare the nature of this invention and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement:—

The present invention relates to small hydraulic pumps and motors intended primarily for use on aircraft for operating various devices and mechanisms thereon, although they may be used for any other purposes for which they may be found useful.

In the case of aircraft, for example, hydraulic pumps are used for creating hydraulic pressure for operating the rams of hydraulic jacks or hydraulic motors for actuating retractable under-carriages, flaps, gun-turrets, and the like, whilst hydraulic motors are used for operating various mechanisms associated with retractable under-carriages, flaps, gun-turrets, variable pitch air-screws, and so on.

The object is to provide a highly efficient motor or pump which operates at very high pressure, and the invention is more particularly concerned with pumps and motors of the swash-plate type, in which the cylinders and pistons are arranged axially in a circle about the driving shaft, with the ends of the pistons bearing against or connected to the swash-plate. In the improved pump according to the present invention, however, the swash-plate is floating i.e. is freely rotatable in relation to both the shaft and the pistons.

In hydraulic motors or pumps of the type indicated above it has been proposed to control the flow of fluid by a valve disc mounted on an extension of the shaft passing through the cylinder block, said valve disc having grooves in one face communicating by internal passages with ports on the other face, and the valve casing having grooves at the same radius as said ports, communicating with the fluid inlet and outlet openings of the casing.

The invention has for its object to provide a generally improved construction of hydraulic motor or pump, and according thereto the driving shaft, which extends through the cylinder block, carries a ported valve disc controlling the passage of working fluid from the valve casing to and from the rear ends of the cylinders, and in front of the cylinder block carries a floating swash-plate which is operatively associated with the projecting ends of the pistons, but is freely rotatable in relation to both the shaft and the pistons.

The invention also includes certain improvements in the construction and arrangement of the valve disc, in accordance with which the valve disc has a continuous peripheral groove communicating with the fluid inlet or outlet of the valve casing and connected by an internal passage in the valve disc with an arcuate groove on the inner surface of the disc contacting with the cylinder heads which have ports located at the same radius as said groove.

According to one construction the valve disc has on its inner surface, another arcuate groove at the same radius as that communicating with the peripheral groove, communicating by an internal passage with a port on the outer surface of the disc, at the same radius as a groove in the valve disc casing communicating with the fluid inlet or outlet of the casing.

According to another construction the valve disc has on its inner surface another arcuate groove at a radius different from that communicating with the peripheral groove, communicating by an internal passage with a port on the outer surface of the disc, at the same radius as a groove in the valve disc casing, communicating with a fluid inlet or outlet, whilst the cylinders have separate inlet and outlet ports at different radii corresponding to the two grooves.

The arcuate extent and relative positions of the two grooves in the valve disc determine the valve timing, which is such that the inlet arcuate groove is over the cylinder ports of the cylinders whose pistons are performing a suction or power stoke, whilst the other arcuate groove is

over the cylinder ports of the cylinders performing an exhaust or delivery stroke, the connections of the various cylinders to inlet and outlet constantly progressing round the circle of cylinders as the swash-plate rotates. Each cylinder will be in communication with the inlet and outlet connections respectively during the whole of its piston travel, thus contributing to high volumetric efficiency.

Various modifications of construction are desirable according as to whether the machine is to be used as a pump or as a motor, and these modifications will now be briefly indicated.

In the case of a pump the inlet and outlet grooves or slots in the valve disc will both have the same pitch circle corresponding to that of the ports of the cylinders which serve both for inlet and outlet, whilst to prevent overheating during long periods of blow-off pressure a fluid relief valve is housed in the end cover, to permit relief of the fluid to a connection on the end-cover, from which it can be conducted to the supply tank or container.

Owing to the high delivery pressure which is desirable, and in order to obtain high volumetric efficiency, it is important that the working clearances between the valve disc, cylinder end-wall, and the end-cover be reduced to a minimum, but in order to minimise surface friction parts of the contact surfaces whch are not subject to delivery pressure, for instance the area on one face of the valve disc between the inlet groove and the shaft, and the whole centre area of the end-cover adjacent to the inlet groove, are preferably relieved.

This relieving renders it possible to provide in a simple manner for automatic lubrication of the complete working parts of the pump. The lubricant can flow from the relieved area of the end-cover to a drilling in the shaft along which it flows to the centre of the pump. By providing a spiral groove in the surface of the driving shaft the lubricant can be returned towards the valve disc, to the relieved area thereof, thus completing the circuit.

In constructing a pump or motor according to the invention the valve casing may be formed by a cover piece secured to the cylinder block, with a distance ring surrounding the valve disc, but according to a modification, instead of using a separate distance ring surrounding and enclosing the valve disc, the distance piece may be integral with the end-cover of the pump. It will be understood, of course, that the joints between the distance piece, the cylinder end-wall, and the end-cover, must make a tight seal, and for this purpose are preferably lapped. Also these parts are located in relation to one another by dowelling and secured as one piece to the cylinder block by bolts.

In the case of a motor the valve disc may be substantially the same as for a pump, if rotation will only be required in one direction, but to permit rotation in either direction the grooving of the valve disc and the porting arrangements must be modified, as hereinafter, explained, owing to the solid areas which must necessarily exist between the ends of the arcuate inlet and outlet slots on the cylinder side of the valve disc, and which serve to provide the necessary seals between the inlet and outlet grooves, between which a considerable pressure difference exists.

These sealing areas shut off each cylinder from both inlet and outlet ports at the two ends of the stroke, but owing to the non-compressibility of the fluid it is essential that each cylinder be in communication with the delivery connection at the commencement of the delivery stroke in the case of a pump, or with the exhaust connection at the commencement of the exhaust stroke in the case of a motor.

In the case of a pump the piston load on the swash-plate is in one direction on the delivery stroke and in the reverse direction on the suction stroke, whilst in the case of a motor the piston load for both the power and exhaust strokes is in the same direction. Consequently the foregoing requirement may be met, in the case of a pump, by providing for a lost motion connection between the pistons and the swash-plate, which allows a sufficient angular movement of the valve disc to permit the sealing areas to pass the cylinder ports whilst the pistons are at their dead centre positions.

An alternative method in the case of a pump is to make the delivery groove in the valve disc a full 180° arc, and to reduce the arc of the suction groove to provide sufficient intervening sealing areas, so that inlet or suction would not continue throughout the whole of the suction stroke. In the case of a non-reversible motor a similar arrangement may be used, the 180° groove being the exhaust groove. Such an arrangement would, however, not be suitable for a reversible motor in which the ports must act alternatively as both inlet and exhaust ports, according to the direction of rotation.

In the case of a reversible motor, therefore, in order to prevent attempted com-

pression of the fluid, the grooves in the valve disc may be arranged at different pitch circles, permitting each groove to be a full 180° arc, with radial sealing areas separating their ends. This involves providing each cylinder with two ports radially spaced correspondingly to the grooves in the valve disc.

Alternatively, instead of having grooves in the valve disc at different pitch circles they may be of equal arcuate extent (somewhat less than 180°) and the cylinders may be provided with a pair of ports spaced circumferentially as far apart as possible, the solid areas between the ends of the grooves in the valve disc being substantially of the same width as the distance between the cylinder ports.

The invention will be more clearly appreciated from the following description, with reference to the accompanying drawings, of constructional embodiments thereof, which are however, only given by way of example, and should not be regarded in a limiting sense.

In the drawings:—

Figure 1 is a longitudinal section on line E—E of Figure 3, of a hydraulic pump constructed in accordance with the invention.

Figure 2 is an end elevation of the driving end of the pump.

Figure 3 is an end elevation of the pump from the opposite end.

Figure 4 is a cross section on the line A—A of Figure 1.

Figure 5 is a cross section on the line B—B of Figure 1.

Figure 6 is a cross section on the line C—C of Figure 1.

Figure 7 is a detail sectional view through the relief valve on the line F—F of Figure 3.

Figures 8, 9 and 10 are a transverse sectional view, and opposite face views, respectively, of the rotary disc valve.

Figure 11 is a longitudinal section of a non-reversible hydraulic motor in accordance with the invention.

Figure 12 is a cross section on the line G—G of Figure 11.

Figures 13, 14 and 15 are a transverse sectional view, and opposite face views, respectively, of the rotary disc valve.

Figures 16 and 17 are diagrammatic and sectional views, respectively, illustrating a modified construction for the rotary disc valve, in the case of reversible hydraulic motors.

Figure 18 is an end view of another modified construction of rotary disc valve in the case of reversible hydraulic motors.

Figure 19 is a partial sectional view illustrating a modified construction of the swash-plate mounting, and

Figure 20 is another partial sectional view illustrating another modification of the swash-plate mounting.

Throughout the drawings corresponding parts of both pumps and motors are indicated by the same reference numerals.

Referring firstly to Figures 1—10, of the drawings, which illustrate a hydraulic pump constructed in accordance with the invention, the pump comprises a body portion, or cylinder block 1, having a central bore 2 for the driving shaft 3, and, arranged in a circle concentric with the bore 2, seven parallel cylinder bores 4 running in the axial direction. The shaft 3 runs in a double purpose ball-bearing 5 carried in a thrust housing 6 bolted over one end of the cylinder block 1, and carrying a mounting flange 7. Projecting from the thrust housing 6 is a housing 8 containing an oil seal of the dual type, held in place by a retaining ring 9. The projecting end of the shaft 3 is splined at 11 for connection to a suitable flexible or other driving or power transmission shaft.

Between the ends of the cylinders 4 and the bearing 5 the shaft 3 is machined cylindrically about an axis inclined to the axis of the shaft to provide an inner race 12 for a double ball-bearing 13, the outer race 14 of which is grooved internally to provide tracks for the balls, and thus acts as an annular swash-plate freely rotatable on the shaft 3. Preferably the swash-plate 14 carries a peripheral rib 15 which is engaged in slots or recesses in the sides of the projecting ends of the pistons 16. As the shaft 3 rotates the swash-plate ring 14 will oscillate angularly in relation to the axis of the shaft, in synchronism with the rotation of the shaft, and will thus reciprocate the pistons 16 cyclically in rotation, driving them positively on both inward and outward strokes. At the same time, due to the ball-bearing 13, the annular swash-plate 14 will not rotate with the shaft, but will remain stationary except for a slight rotational creep due to the inevitable but small amount of friction arising from the ball-bearing 13, which will, however, be balanced to a large extent by the frictional drag of the rib 15 in the recesses in the pistons 16.

At the opposite end of the cylinder block 1 the ends of the cylinders are closed by a cylinder head-plate 17 in which are holes 18 opening by flared passages into the cylinders 4. Adjacent to the cylinder head-plate 17 and running in close contact therewith is mounted on the shaft 3 the rotary disc valve 19,

shown in detail in Figures 8, 9 and 10, which is surrounded and enclosed by an annular distance piece 20, which may be a separate ring member as in Figure 1, or integral with the end-cylinder block or with the cover 21 of the pump or motor as illustrated in Figure 11. The end-cover 21 carries the fluid inlet connection 22 and the fluid outlet connection 23, and also a spring pressed ball relief valve 24, which permits relief of pressure fluid during long blow-off periods from the outlet connection 23 to a connection 25 leading to the fluid supply tank or container, thereby preventing overheating. The distance piece 20, end-cover 21, cylinder head-plate 17, and cylinder block 1 are provided with lugs by which they are fastened together by securing bolts 26, the three former being preferably registered by dowelling and secured as one piece to the cylinder block 1. The joints are lapped in the finishing process so as to ensure a fluid tight seal.

The rotary valve disc 19 has on the face adjacent to the cylinder head-plate 17, a pair of arcuate grooves or recesses pitched at the same radius as the ports 18 in the cylinder head-plate 17. These grooves are sealed from one another by the intervening solid areas of the disc 19. The groove 27, which may be termed the inlet or suction groove, is connected by a passage through the valve disc 19 with a port 28 in the outer face of the disc which is at the same radius as a groove 29 in the adjacent face of the end-cover 21, which is constantly in communication with the inlet connection 22. The groove 30, which may be termed the outlet or delivery groove, is connected by a radial passage 31 within the thickness of the valve disc 19 with a continuous peripheral groove 32 round the edge of the valve disc. This groove 32 is continually in communication through the passage 33 in the distance piece 20, with the outlet connection 23 of the end-cover 21. Consequently as the valve disc 19 rotates with the shaft 3 the cylinders 4 are periodically brought by the inlet groove 27 into communication with the inlet connection 22 and by the outlet groove 30, into communication with the outlet connection 23, and by suitably mounting the valve disc 19 on the shaft 3 its operation may be so timed that each cylinder is brought and maintained in connection with the inlet connection 22 during the whole of its outward or suction stroke, and in connection with the outlet connection 23 during the whole of its inward or delivery stroke. This results in high volumetric efficiency, and with seven cylinders there will be never less than three cylinders delivering, so that there will be little or no pulsation at any speed or fluid pressure.

Owing to the solid portions of the valve disc 19 between the ends of the grooves 27 and 30, there will be moments at the ends of their strokes when the cylinders 4 are cut off from both inlet and outlet, and in order to prevent any possibility of attempted compression of the fluid in the cylinders by small movements of the pistons taking place during these periods, the recesses or slots in the sides of the pistons 16 may be made slightly wider than the rib 15 on the swash-plate ring 14, so that there is a sufficient degree of lost motion to enable the valve disc 19 to perform a small angular movement without causing corresponding movement of the pistons 16. If this expedient is adapted the inlet and outlet grooves 27 and 30 may be of equal lengths approximating each to $^2/_5$ of the pitch circle, the remaining $^1/_5$ of the circle being divided between the two solid areas between the ends of the grooves.

Another expedient, as illustrated by Figures 8, 9 and 10, may, however, be adopted, namely, that of making the outlet groove 30 occupy substantially 180° of the pitch circle, and making the inlet groove 27 correspondingly shorter to provide the two solid areas between the ends of the grooves. With this construction, however, the cylinders would not be connected to the inlet connection 22 for the whole period of the suction stroke.

Due to the high working pressure of the fluid and in order to obtain high volumetric efficiency it is important that the working clearances between the valve disc 19, cylinder head-plate 17, and end-cover 21, be kept down to a minimum, but on the other hand in order to reduce frictional losses surface contact between the parts is undesirable. Consequently parts of the surface of the valve disc 17, which are not subjected to the high delivery pressure and are not acting as a seal, are preferable relieved. Thus the radial area 34 between the inlet groove 27 and the centre of the disc, adjacent to the cylinder head-plate 17, and the whole annular centre area 35 within the inlet groove 29 in the end-cover 21, may be relieved.

This relieving of surfaces renders possible an automatic system of lubrication throughout the working parts of the pump, as will be seen from the drawings. Part of the working fluid will pass from the relieved area 35 in the end-cover 21 to a bore 36 along the shaft 3 and thence by a radial bore 37 into the chamber containing the swash-plate. It will then travel backwards towards the valve disc

19 along the spiral groove 38 in the surface of the shaft 3, to the relieved area 31 of the valve disc 19, whence it will be sucked up by the cylinders 4, thus completing the circuit.

Referring now to Figures 11—15, illustrating a construction of motor according to the invention, the motor illustrated is a non-reversible motor of a construction substantially the same as that of the pump already described, except as hereinafter explained, it being understood that the terms "inlet" or "suction" as applied to the pump correspond to "inlet" as applied to a motor, and the terms "outlet" or "delivery" as applied to the pump correspond to "outlet" or "exhaust" as applied to a motor.

The main difference between the pump and motor, as illustrated, is that the motor contains a greater number of cylinders, for example nine, to give a more even torque and the pistons have a longer stroke resulting from a greater inclination of the swash-plate in relation to the axis of the shaft 3. The relief valve 24 and its casing on the end-cover 21 are omitted as being unnecessary.

The swash-plate ring 14 is also journalled in a different manner on the shaft 3, in order to take the greater end thrust, by means of a single ball-ring 39 adjacent to one edge of the swash-plate ring 14, and a taper roller bearing 40 for the other edge of the swash-plate ring. Consequent upon the greater weight of the swash-plate assembly and its greater stroke it is desirable to provide on the shaft a balance weight 41 of suitable weight according to the design, the weight increasing with the stroke of the swash-plate. The ball thrust bearing 5 of the pump is replaced by a double purpose taper roller bearing 42 held in the thrust housing 6.

Since the pistons 16 of the motor will only be subject to a uni-directional load, it is not necessary for them to have a positive connection with the swash-plate ring 14, and accordingly their ends may simply be rounded to make frictional contact with the surface of the swash-plate 14. Certain other modifications of construction are, however, necessary when the motor is to be reversible, principally in regard to the passages and ports in the valve disc 19. However, it is also not possible to employ the automatic lubrication system described for the pump, owing to the absence of relieved areas on the valve disc and end-cover, since the passages and ports of a reversible motor must serve alternatively for high and low pressure fluid, and relieved surfaces cannot be made use of adjacent to high pressure fluid passages. In this case therefore, the interior of the motor containing the swash-plate mechanism may be maintained full of oil, through a pipe indicated in dotted lines at 43, connected to an oil supply tank or container. There will be no rise in pressure due to any leakage past the pistons, since the fluid will have free passage to the supply tank.

As regards the ports and passages in the valve disc 19, special care must be taken to avoid the risk of the pistons attempting to compress the working fluid in either direction of running, i.e. the pistons starting on an exhaust stroke before the cylinders are open to exhaust. It follows from this that the grooves 30 and 27 of the valve disc 19 must be of equal angular extent, since they have to serve alternatively for both inlet and exhaust. A convenient arrangement to meet this requirement is illustrated in Figures 16 and 17, in which the grooves 27 and 30 in the valve disc 19 are arranged at different pitch circles, and the cylinder head-plate 17 is provided with a pair of ports 18*a* and 18*b* for each cylinder 4 at different radii corresponding to the two grooves. By this means the two grooves 27 and 30 may be of equal angular extent viz a full 180°, so that no cylinder is completely shut off from either inlet or exhaust for more than an infinitesimal period. In this case the ends of the grooves are sealed from one another by a radially extending area of the disc 19, the width of which may be increased within limits, by spacing the ports 18*a* and 18*b* further apart than the diameter of the cylinder 4 and connecting them to the cylinders by inclined passages 18*c*.

An alternative arrangement is illustrated by Figure 18, in which each cylinder 4 again has two ports 18*a* and 18*b*, but they are circumferentially spaced as far apart as may be necessary, in the same pitch circle, to establish a satisfactory seal between the ends of the grooves 27 and 30, which in this case are at the same pitch circle corresponding to that of the ports 18*a* and 18*b*. In this case, of course, the two grooves 27 and 30 will be of equal lengths but slightly less than 180° arc, corresponding to the width of the sealing gaps between the ends of the grooves.

In both of these cases employing two ports 18*a*, 18*b* for each of the cylinders 4, the inlet will be momentarily connected to the exhaust via the cylinder ports and valve disc 19, but this will be for so short a period of time as not to

affect the efficiency of the motor to any appreciable extent.

Figure 19 shows a modified construction of the swash-plate mechanism for a pump, which is similar to that shown in Figure 11, for a motor, inasmuch as one of the ball-bearings 13 of the swash-plate ring 14 and the ball-bearing 5 of the shaft 3 are replaced by taper roller-bearings corresponding to 40 and 42 of Figure 11. In addition a balance weight 41 may be provided.

Figure 20 shows another modification for the mounting of the swash-plate ring 14, for either pumps or motors, which is similar to that of Figures 11 and 19 except that the ball bearing 39 is replaced by a plain bearing 44.

From the foregoing it will be appreciated that many modifications may be made in the constructional details of both pumps and motors without departing from the invention.

Having now particularly described and ascertained the nature of my said invention and in what manner the same is to be performed, I declare that what I claim is:—

1. A hydraulic motor or pump of the type set forth in which the driving shaft, which extends through the cylinder block, carries a ported valve disc controlling the passage of working fluid from the valve casing, to and from the rear ends of the cylinders, and in front of the cylinder block carries a floating swash-plate which is operatively associated with the projecting ends of the pistons, but is freely rotatable in relation to both the shaft and the pistons.

2. A hydraulic motor or pump according to claim 1, in which the valve disc has a continuous peripheral groove communicating with the fluid inlet or outlet of the valve casing and connected by an internal passage in the valve disc with an arcuate groove on the inner surface of the disc contacting with the cylinder heads which have ports located at the same radius as said groove.

3. A hydraulic motor or pump according to claim 2, wherein the valve disc has on its inner surface, another arcuate groove at the same radius as that communicating with the peripheral groove, communicating by an internal passage with a port on the outer surface of the disc, at the same radius as a groove in the valve disc casing communicating with the fluid inlet or outlet of the casing.

4. A hydraulic motor or pump according to claim 2, wherein the valve disc has on its inner surface another arcuate groove at a radius different from that communicating with the peripheral groove, communicating by an internal passage with a port on the outer surface of the disc, at the same radius as a groove in the valve disc casing, communicating with a fluid inlet or outlet, whilst the cylinders have separate inlet and outlet ports at different radii corresponding to the two grooves.

5. A hydraulic motor or pump according to claim 3, wherein the arcuate groove on the inner face of the valve disc communicating with the peripheral groove is of greater length than the second arcuate groove.

6. A hydraulic motor or pump according to claim 4, wherein the two arcuate grooves on the inner face of the valve disc are of equal lengths.

7. A hydraulic pump according to any of the preceding claims, wherein the end-cover of the valve casing is provided with a fluid relief valve permitting passage of fluid from the outlet connection to the supply tank or fluid container during long periods of " blow-off " pressure.

8. A hydraulic motor or pump according to any of claims 1 to 6, or a pump according to claim 7, wherein areas of the valve disc which are not subject to inlet or delivery pressure, as the case may be, are relieved to reduce surface friction between the disc and the adjacent surfaces of cylinder block and end-cover.

9. A hydraulic pump or motor according to claim 8, wherein the central area within the inlet groove on the surface of the end-cover, and a radial section within the inlet groove on the valve disc, are relieved.

10. A hydraulic pump or motor according to claim 9, having an automatic system of lubrication in which part of the working fluid is circulated from the relieved area of the end-cover, through a bore in the shaft running longitudinally and opening into the swash-plate chamber, and back to the relieved area of the valve disc along a spiral groove in the surface of the shaft.

11. A hydraulic pump or motor according to claim 5, wherein the second arcuate groove in the valve disc through which the inlet or suction fluid passes is less than 180° of arc, and the groove through which the fluid passes to the delivery or exhaust connection is a full 180° of arc.

12. A reversible motor according to claims 4 and 6, wherein each cylinder has a pair of ports in radially spaced relation and the two arcuate grooves in the valve disc are at different pitch circles corresponding to the two sets of cylinder ports, and both of substantially 180° of arc.

13. A reversible motor according to claims 4 and 6, wherein each cylinder has

a pair of ports circumferentially spaced round the same pitch circle, and the two arcuate grooves in the valve disc are at the same pitch circle corresponding to that of the cylinder ports, and both of equal arc somewhat less than 180°.

14. A reversible motor according to any of the preceding claims, which is maintained full of lubricating oil through a pipe connected to the swash-plate chamber of the motor.

15. A hydraulic pump or motor of the type set forth, comprising a cylinder block centrally through which passes the driving or driven shaft, a plurality of axial cylinder bores disposed round a pitch circle concentric with said shaft, a cover plate secured to the front end of said cylinder block, which supports a bearing for said shaft and forms a chamber containing a floating swash-plate freely rotatable on said shaft, pistons in said cylinders operating, or operated by said swash-plate, which is freely rotatable in regard to said pistons, a cylinder head-plate on the other end of said cylinder block and having ports communicating with said cylinders, an end-cover carrying fluid inlet and outlet connections which is secured to said cylinder block and is spaced from said cylinder head-plate by a distance ring, a rotary valve disc carried by said shaft in the space enclosed by said distance ring, and passages and ports in said end-cover and valve disc, whereby fluid is enabled to enter and leave said cylinders in timed relation to the movement of said pistons.

16. Hydraulic pumps substantially as herein described with reference to, and as illustrated by Figures 1—10 of the accompanying drawings, with or without the modifications referred to.

17. Hydraulic motors substantially as herein described with reference to and as illustrated by Figures 11—18 of the accompanying drawings, with or without the modifications referred to.

Dated this 6th day of July, 1938.

CHARLES S. PARSONS,
Chartered Patent Agent,
Thanet House,
231, Strand, London, W.C.2,
Agent for the Applicant.

Leamington Spa: Printed for His Majesty's Stationery Office, by the Courier Press.—1940.

Fig. 1.

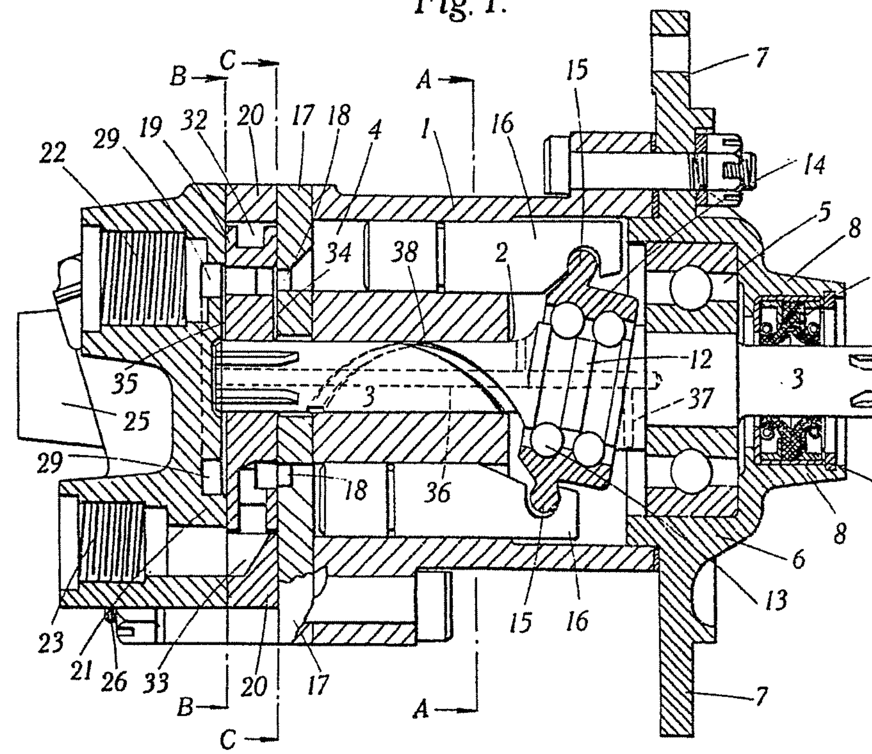
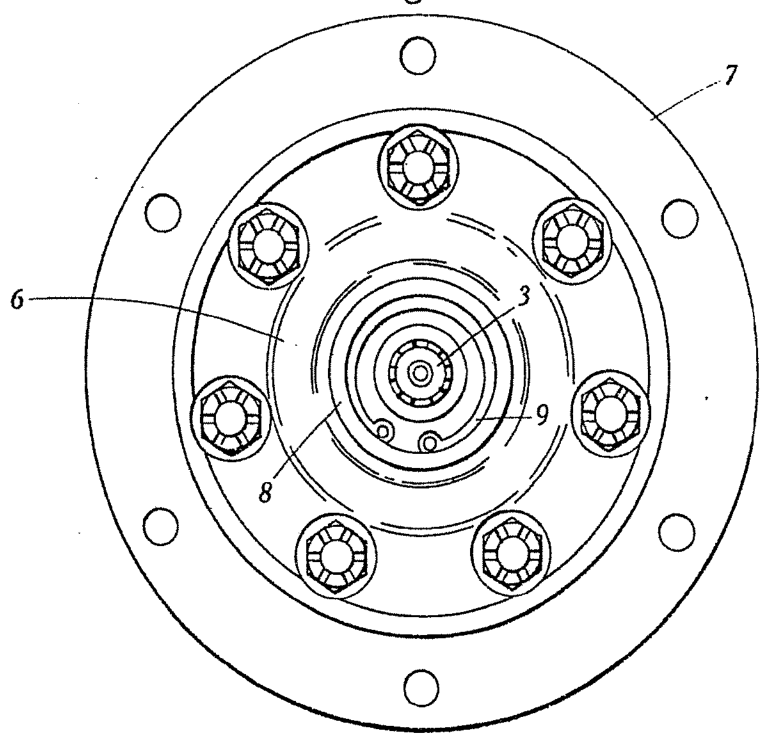


Fig. 2.



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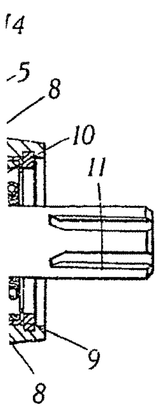
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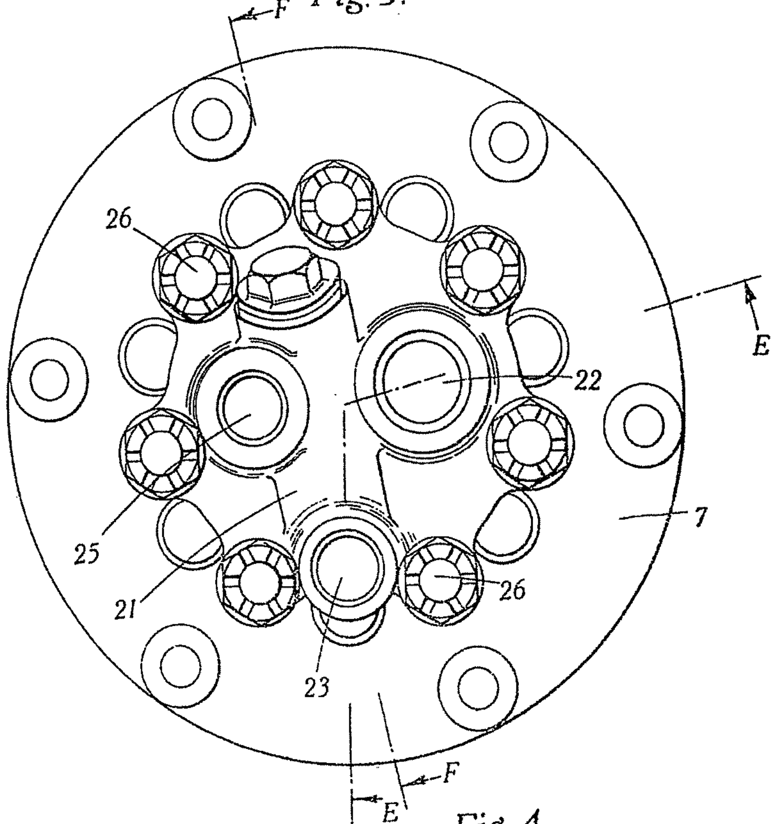
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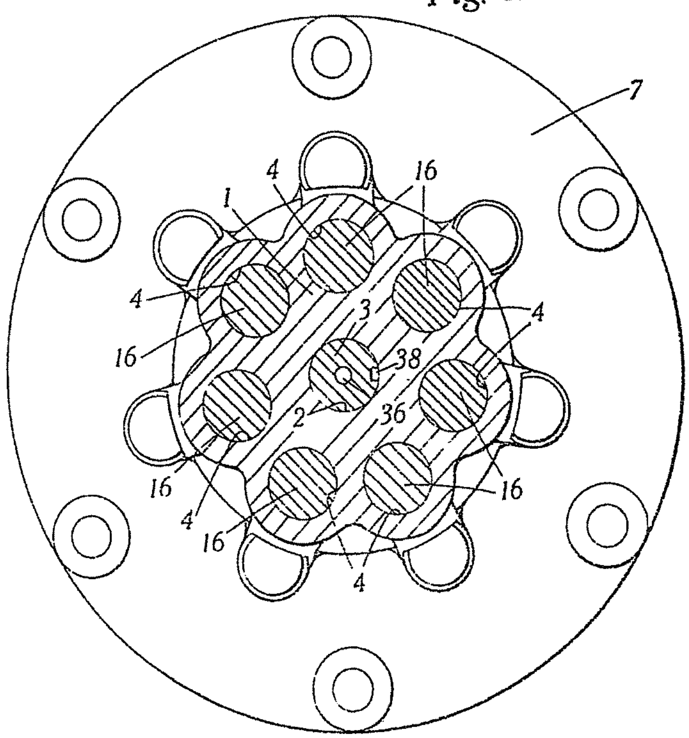
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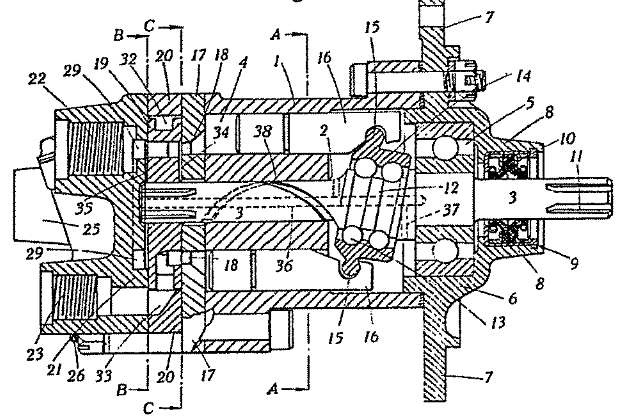
515,635 COMPLETE SPECIFICATION

5 SH
SHEET

SHEET 1

Fig. 1.



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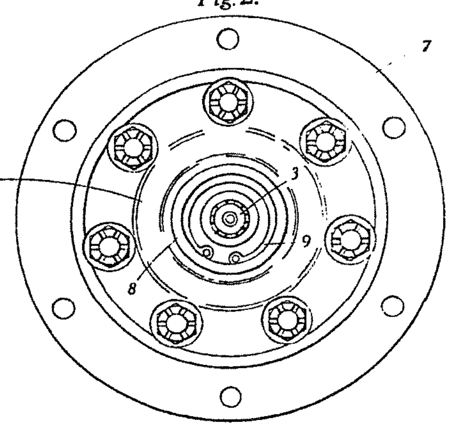

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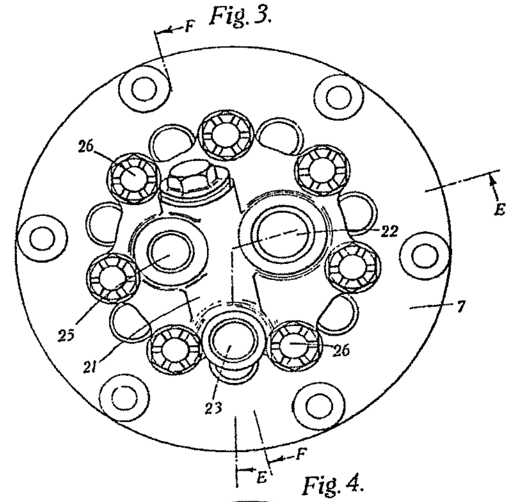

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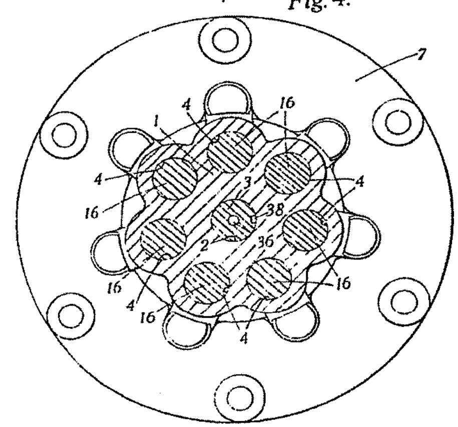

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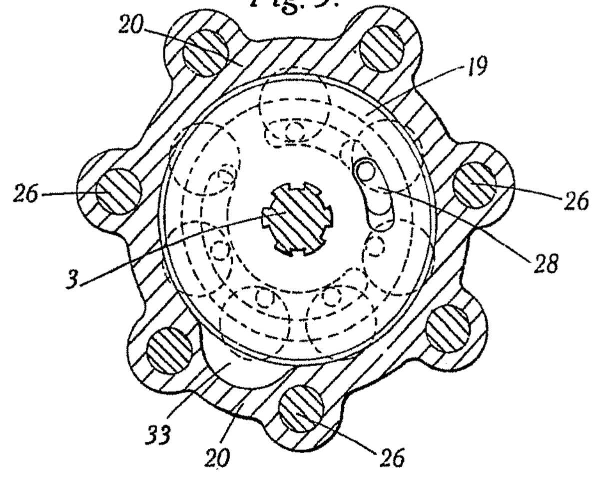
Fig. 5.



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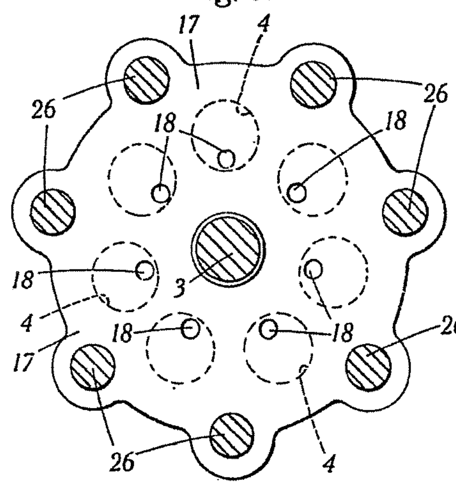

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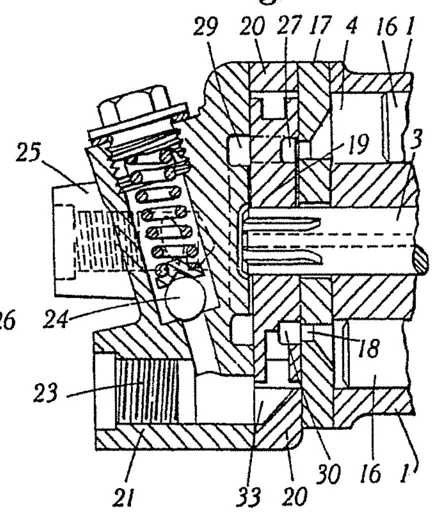

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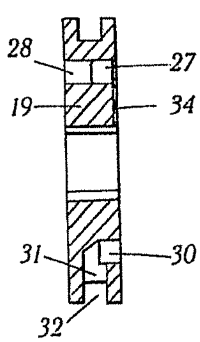

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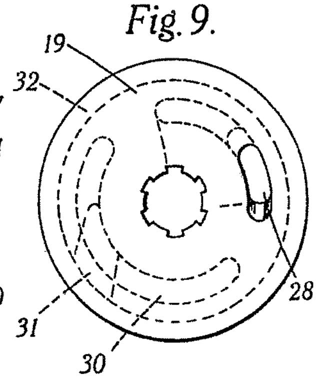

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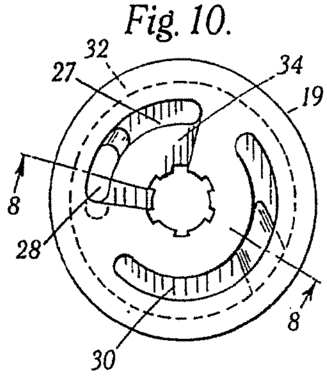

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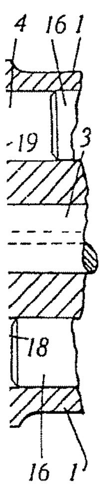

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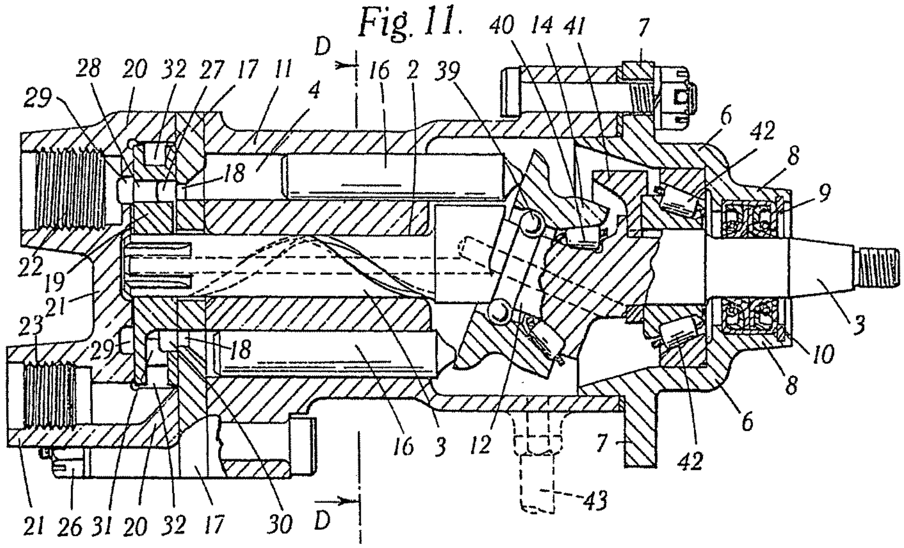

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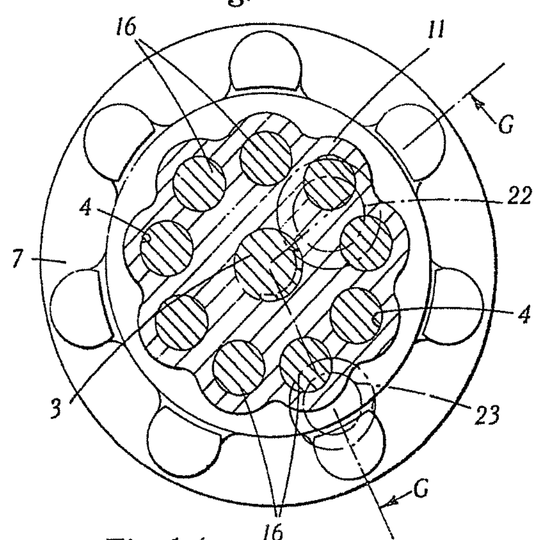

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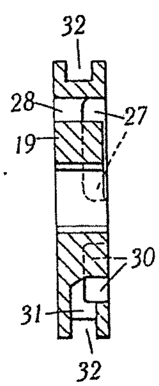

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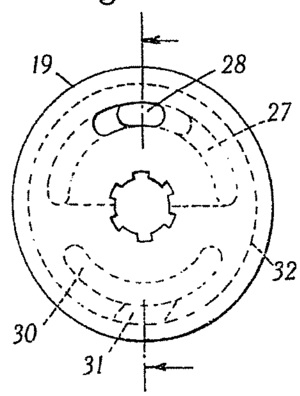

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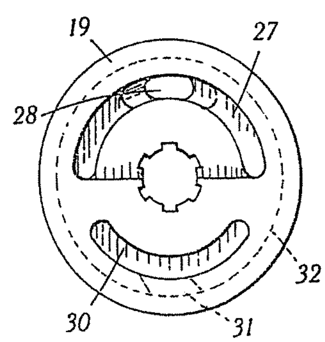

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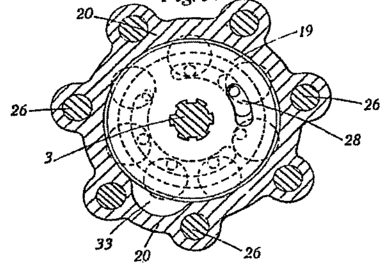

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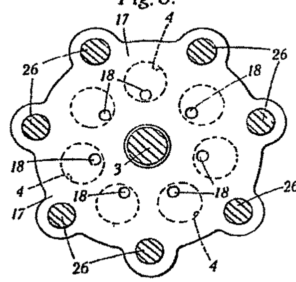

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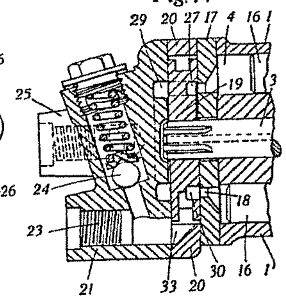

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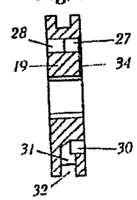

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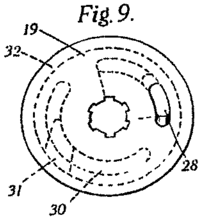

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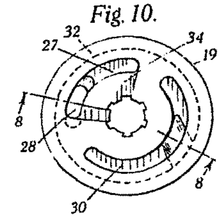

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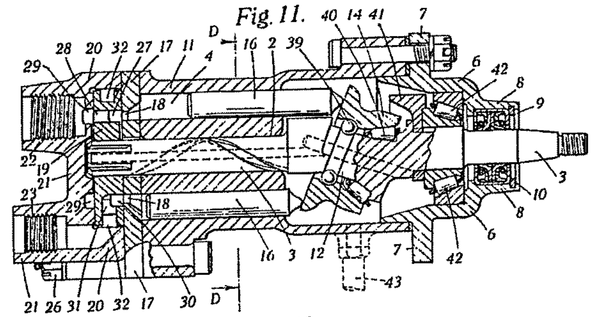

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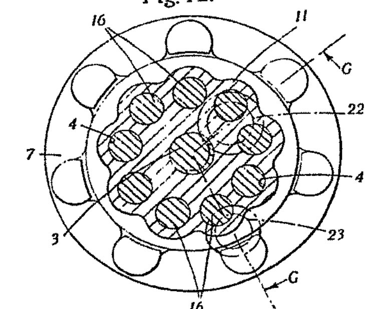

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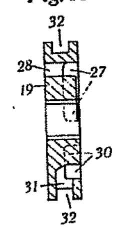

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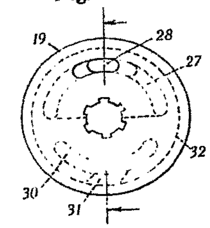

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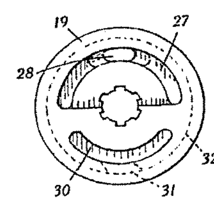

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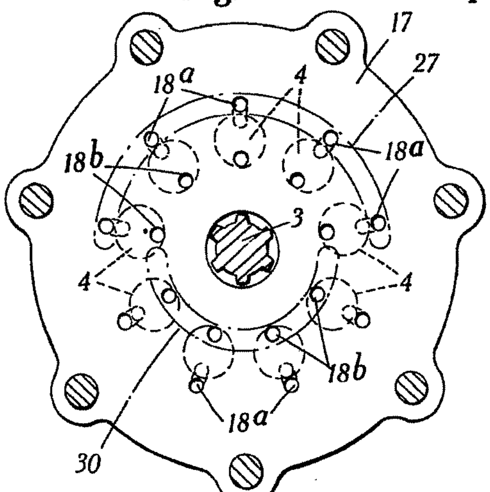

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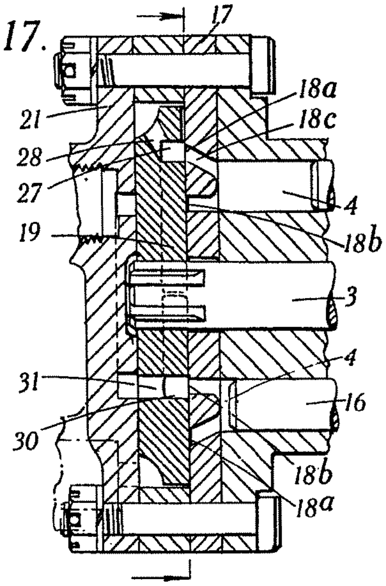

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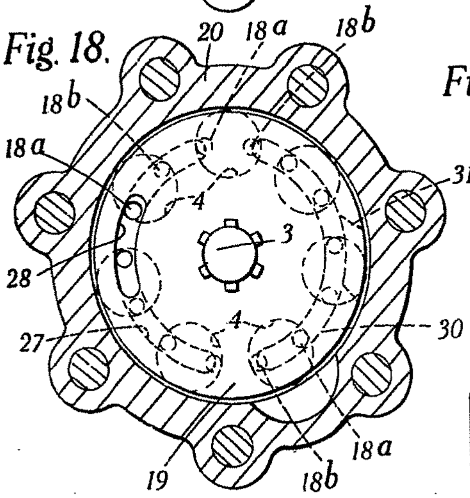

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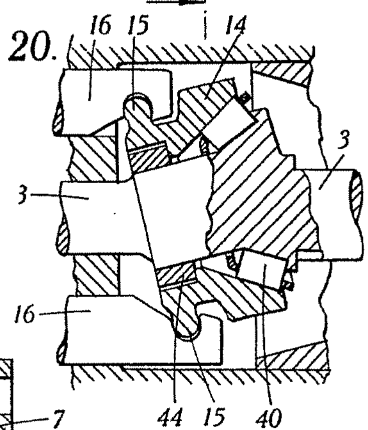

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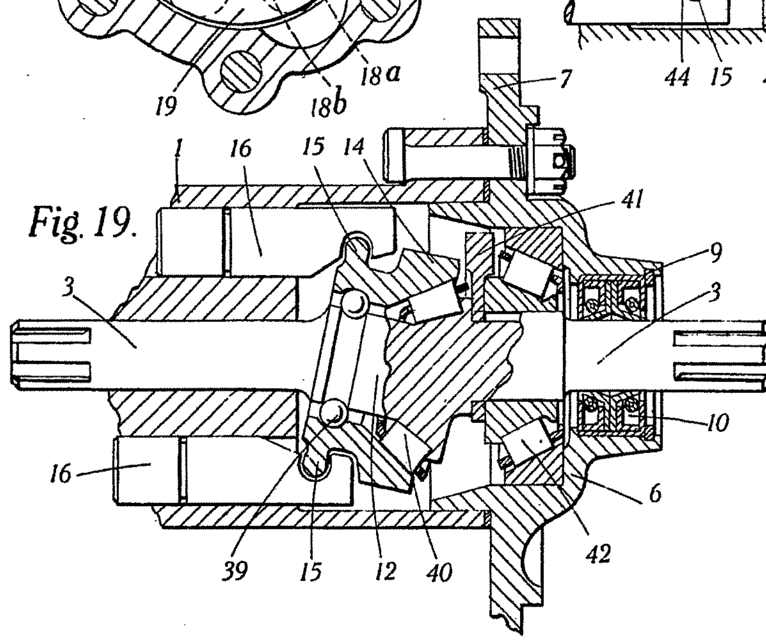

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